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Selection of Labyrinth Seals in Steam Turbines

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Abstract—The efficiency, vibration stability, operational durability, and cost of the main types of peripheral seals used in steam turbines are considered. A comparison between the conventional and honeycomb seals is given. Conditions subject to which replacement of conventional seals by honeycomb ones can be justified are pointed out. The use of variable-pitch multicomb seals as the most promising ones is recommended.

Keywords: labyrinth seals, leak losses, exciting forces, shaft system vibration stability, durability of seals, cost of seals

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Decreasing peripheral, diaphragm, end, and intermediate leaks in the steam turbine flow path is an important component of efforts taken at improving the turbine inner efficiency. One typical solution of this task involves the use of labyrinth seals.

As a rule, peripheral leaks exert the strongest influence on the turbine efficiency, the losses from which are usually a few (from 2 to 5) times larger than those from other leaks. Therefore, the main attention should be paid to peripheral labyrinth seals.

Comparison among different types of labyrinth seals should be carried out with respect to the following characteristics: *flow rate, dynamic, operational durability, repairability, and cost.* In the present study, we compare the types of peripheral seals shown in the Figs. 1a–1d.

Figure 1a shows the typical peripheral seals that were widely used in almost all domestically produced turbines in their high- and intermediate-pressure parts (HPP and IPP), as well as (with some modifications) in the low-pressure part (LPP) till the late 1960s.

In the early 1970s, the problem of self-exciting low-frequency vibration was encountered in mastering the manufacture and use of the T-250/300-23.5 turbine produced by the Ural Turbine Works (UTZ). To solve this problem, engineers of the UTZ (the then Ural Turbine Engine Works, UTMZ) elaborated the design of axial-radial seals (see Fig. 1b), the philosophy of which had been suggested by engineers from the Moscow Power Engineering Institute (MEI). The MEI–UTMZ axial-radial seals had quite large radial gaps δ_1 and moderate axial gaps δ_3, which just offered the main resistance to leaks through seals.

In view of the fact that the per unit length leak through the axial-radial seals is almost independent on the radial displacement of the rotor in the stator bore, the bucket force and the slit component of the overshroud exciting force are close to zero, and only the channel component remains different from zero. Assessments show that in comparison with the conventional seal (see Fig. 1a), the exciting force in the axial-radial seal is smaller (for the control stage used in a K-300-23.5 turbine) by more than a factor of 5 [1].

Below, the above-mentioned types of seals are compared with respect to their main characteristics.

Seal efficiency. Losses in a stage due to peripheral leak ζ_p serve as a measure of seal efficiency. The loss calculation procedure for all types of seals is described in [2]. The general formula for calculating ζ_p has the following form:

$$\zeta_p = \frac{1}{\sin\alpha_1}\left(\rho_m + 1.7\frac{l_1}{d}\right)^{1/2}\left(1 + k_{p.l}\right)\frac{\delta_{eff}}{l_1}\left(\frac{d_p}{d}\right), \qquad (1)$$

where α_1 is the flow exit angle from the nozzles at the mean diameter, ρ_m is the stage reaction degree at the mean diameter, d is the nozzle bucket mean diameter, $k_{p.l}$ is the coefficient that takes into account the effect from admixing the peripheral leak to the main flow, δ_{eff} is the effective gap in the peripheral seal, and d_p is the runner peripheral diameter.

The influence of ζ_p on the efficiency taking peripheral leak into account is determined from the dependence

$$\eta'_{r.i} = \eta_{r.b}\left(1 - \zeta_p\right), \qquad (2)$$

where $\eta_{r.b}$ is the relative blade efficiency of the stage.

Assuming that the comparison is carried out for one particular stage, for which the LMZ K-300-23.5 turbine's HPP sixth stage is taken, we will consider that all parameters for four cases are the same and differ only in the value of effective gap δ_{eff}.

The common parameters in the stages being compared have the following values: $l_1 = 61$ mm, $\alpha_1 = 14°$,

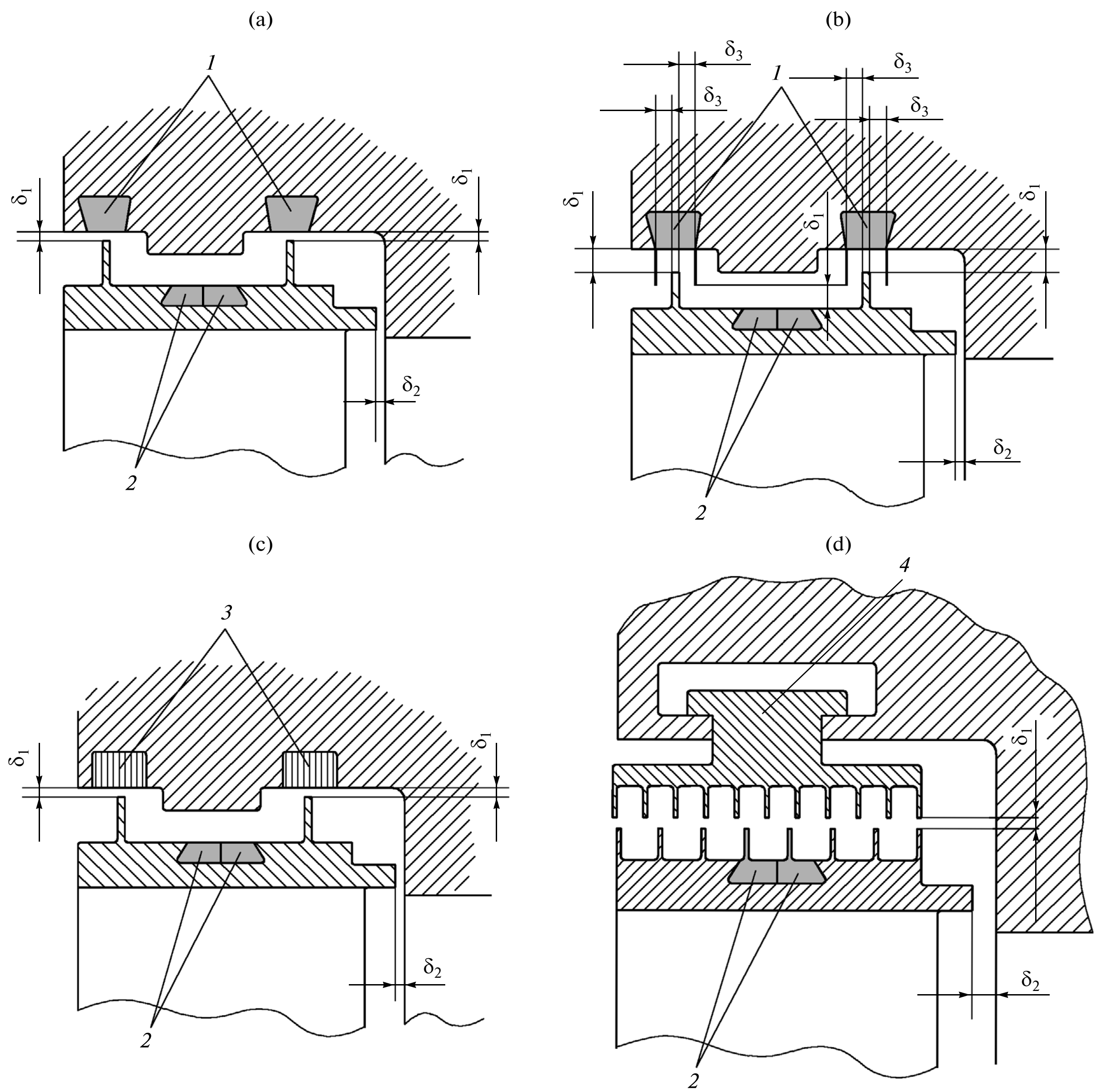

Fig. 1. Compared types of peripheral seals. (a) Conventional, (b) axial-radial, (c) honeycomb, and (d) variable-pitch multicomb ones. (*1*) Cermet inserts, (*2*) damping inserts, (*3*) honeycomb inserts, and (*4*) segments self-aligning in the working position.

$\rho_m = 0.25$, $d = 0.874$ m, $k_{p.l} = 0.1$, $d_p = 0.935$ m, the axial gap $\delta_2 = 2.5$ mm, and the flow rate coefficient for axial seal with a 2.5 mm gap $\mu_2 = 0.47$.

The calculations of δ_{eff} for all cases were carried out according to the procedure described (with examples) in [2]. The following conclusions can be drawn from the calculation results summarized in Table 1:

(i) With the same peripheral gaps $\delta_1 = 1.2$ mm, the losses due to peripheral leaks in the conventional and honeycomb leaks are approximately equal to each other (which should be expected), and the values of efficiencies are the same $\eta'_{r.i} \approx 0.88$.

(ii) With the gaps in honeycomb seals equal to 0.6 mm, the losses due to leaks in them are equal to 2/3 of the losses in the conventional seals with the gap in them $\delta_1 = 1.2$ mm, and, accordingly, the efficiency of the stage with honeycomb seals is by 1% higher than that of the conventional stage.

(iii) In comparison with the stage with honeycomb seals and equal radial gaps $\delta_1 = 1.2$ mm, the stage with variable-pitch multicomb seals (VMSs) has a factor of 2.5 smaller leakage losses and by 2% higher efficiency.

(iv) In comparison with the stage with honeycomb seals at $\delta_1 = 0.6$ mm, the stage with VMSs at $\delta_1 =$

Table 1. Characteristics of the compared stages fitted with different types of seals

Indicator	Seal type					
	conventional	axial-radial	honeycomb		VMS	
Radial gap δ_1, mm	1.2	5.0	0.6	1.2	0.6	1.2
Radial gap flow rate coefficient μ_1	0.725	0.7	0.875	0.815	0.75	0.75
Effective gap δ_{eff}, mm	0.678	0.715	0.452	0.731	0.231	0.289
Peripheral leak loss coefficient ζ_p	0.0328	0.0346	0.0219	0.0354	0.0112	0.0140
Efficiency $\eta'_{r.i}$	0.880	0.878	0.890	0.878	0.900	0.897

The flow rate coefficients μ_1 were determined using the data from [2] and those for honeycomb seals, from the results of test-bench experiments carried out at the MEI.

1.2 mm has a factor of 1.5 smaller leakage losses and by 1% higher efficiency.

Since the efficiency of the HPP sixth stage is almost the same as the average efficiency of the high-frequency part, the values presented in Table 1 give $\eta'_{r.i}$ for the entire HPP of the K-300-23.5 and T-250/300-23.5 turbines.

Thus, a conclusion can be drawn from a comparison of the efficiencies of seals that the use of honeycomb seals in the HPP (in case of decreasing the radial gaps from 1.2 to 0.6 mm) really gives a 1% increase in the efficiency of the HPP used in a 300-MW supercritical-pressure turbine.

The authors who give information in their works about possible increase of the HPP efficiency by 2% (see, for example, [3]) mistake the wish for the reality because for reaching this value, the radial gaps δ_1 must be decreased to 0.3 mm, which is practically impossible. At the same time, in case of using VMSs, increasing the HPP efficiency by 2% is quite realistic.

The dynamic characteristics of seals (the influence on vibration stability). The value of exciting (bucket, overshroud, and labyrinth) forces arising in the stages and provoking self-excited rotor vibration is the measure characterizing the effect of seal on the rotor vibration stability. The exciting forces depend directly on the pressure of medium; therefore, their highest values occur in the control stage, and they also have significant values in the HPP first uncontrolled stages.

Table 2. Expert estimates of the stiffness ratios in stages fitted with different types of seals

Indicator	Seal type				
	conventional	axial-radial	honeycomb		VMS
Radial gap δ_1, mm	1.2	5.0	0.6	1.2	1.2
D/D^{conv}	1.0	0.2	2.0	1.0	0.6

D^{conv} is the stiffness of conventional seals.

The overshroud force arising in conventional seals is inversely proportional to the radial gap value δ_1. According to the calculation results [1], as δ_1 is varied from 1.2 to 0.6 mm, this force increases by approximately a factor of 2.

There are grounds to suppose (direct evidence does not seem to be available) that the general theory of calculating the exciting forces in conventional seals is applicable for honeycomb seals. Then, the values of stiffness D for conventional seals are approximately equal to those for honeycomb seals.

As far as we know, there are no theoretical investigations of exciting forces in VMSs.

A conjecture can be made based on the results from a few experiments [4] that the exciting forces in VMSs at $\delta_1 = 1$ mm are equal to 60% of those characteristic of traditional seals.

In view of what was said above, Table 2 gives the expert estimates of ratios for stiffness D in the stages with the considered seals. It follows from the data presented in the table that the axial-radial seals are absolutely superior ones in the level of exciting forces. *It is important to note that replacement of axial-radial seals by honeycomb ones with $\delta_1 = 0.6$ mm results in a ten-fold growth of exciting forces in the stages.*

Durability of seals in operation. By durability we understood absence of damage in the seals in normal and transient modes of turbine set operation, in particular, under the conditions of increased rotor vibration, in the startup modes, in load variation modes, and in the rotor shutdown and rundown modes, when the axial and radial gaps change (decrease to zero) as compared with their nominal values.

The intactness of seals is upset not only as a consequence of the rotor rubbing against the stator, but also due to some other factors, such as corrosion, mechanical erosion, and electrical erosion. However, these kinds of damage seem to be little dependent on the kind of seal. Therefore, we will confine our consideration to typical kinds of damage inflicted to seals due to rubbing.

One of the negative consequences caused by rubbing is its adverse effect on the turbine efficiency and reliability. Obviously, the larger the axial and radial gaps in the seals, the less probable the occurrence of rubbing in them.

Axial-radial seals are, obviously, the best ones among the considered types of seals in terms of rubbing occurrence conditions (see Fig. 1b) because the axial gaps δ_2 in them are the same as in the other seals, whereas the radial seals δ_1 are the maximal ones (δ_1 = 5 mm) and are essentially larger than in the other seals.

Honeycomb seals are the worst ones with respect to this criterion (see Fig. 1c); the advocates of using these seals recommend radial gaps to be installed at a level of 0.6 mm.

Conventional seals and VMSs, in which both axial δ_2 and radial gaps δ_1 are taken equal to each other, occupy an intermediate position.

In making a comparison between conventional and honeycomb seals, emphasis is placed on the fact that the extent to which stage efficiency decreases due to rubbing in honeycomb seals is smaller than that in conventional seals because the combs locally penetrate into the honeycomb cells. The comb itself is not subjected to fretting, and the slots formed in the cells have an insignificant effect on the leak of medium through the seals. When rubbing occurs in conventional seals with smooth walls, the comb is subjected to wear, the gap δ_1 in the seal increases, and the leak grows in proportion to the gap increase.

However, it should be pointed out that by properly selecting the materials for the comb, shroud, and cermet insert (if such an insert is used), operating conditions can be obtained in which no damage is inflicted to the comb as it penetrates into the cermet in rubbing. In fact, this is the core of the philosophy of using cermet inserts in conventional seals.

Conventional seals—either with or without cermet inserts—are usually installed with radial gaps equal to 1.2–1.5 mm, whereas in case of honeycomb inserts it is commonly accepted than the gaps can be decreased to 0.5–0.6 mm.

Different specialists participating in discussing the problem in question give different estimates to the experience gained from operation of turbines with honeycomb seals. Some of them actively advocate the use of honeycomb seals in steam turbines and emphasize their positive features such as achieving better economic efficiency and durability, whereas the others point out cases of unsuccessful operation and failures that occurred in turbines due to the specific features of the honeycomb seals used in them [5, 6].

More economically efficient operation and better durability of honeycomb seals in case of using them instead of conventional ones can be achieved subject to fulfilling certain conditions.

1. A sufficient vibration stability margin shall be provided in the turbine set shaft system; i.e., the relative threshold steam flow rate must be $\bar{G}_{thr} \geq 1.5$. This condition must be estimated both in designing new turbines and in upgrading old ones involving a shift for using honeycomb seals. There are a procedure and a computer program for determining $\bar{G}_{thr}$ [7], which are available at some manufacturing plants.

2. The rotor alignment requirements during the installation operations must be improved and made more stringent taking into account rotor lift-off on the oil film, temperature deformation of the outer casing, and tightening of the flanges in the horizontal joint of the HPP and IPP outer shell.

3. Operational practices must be improved with a view to minimize the dynamic sags of the rotor in the transient modes of operation, specifically, in the turbine startup and shutdown modes with rotor rundown.

4. Since the passage of shaft system critical frequencies during the startup and especially in the rundown mode without rubbing is a difficult task, a self-adjustment design of seals with increased gaps in transient modes of operation should be used for simplifying and implementing this task. General Electric and Siemens use such seals.

A comparison between variable-pitch multicomb seals and honeycomb seals with respect to all considered parameters shows that VMSs have certain advantages: the HPP has an increased efficiency by approximately 1% (see Table 1), they have better durability in operation due to acceptability of using increased radial gaps, and they are characterized by a lower level of exciting forces (see Table 2): in regard of this parameter, VMSs are close to axial-radial seals and are better than all other seals.

Siemens uses VMSs in some stages of steam turbines. Russian manufacturers of turbine machinery and equipment do not use VMSs. The advantages of VMSs are demonstrated in many publications written by specialists of the MEI Chair for Steam and Gas Turbines [2, 8, 9, and others]. The Turboatom Company widely uses VMSs in the projects of new turbines [10].

The cost of seals is of no small importance. Estimates made by experts testify that honeycomb seals are the most expensive ones among the considered types of seals. As to their popularity, it is stemming from their exaggerated advertisement, which emphasizes their advantages and shades their shortcomings.

CONCLUSIONS

(1) An analysis of the main indicators characterizing the used overshroud seals shows that variable-pitch multicomb seals are the best ones in terms of economic efficiency, and that axial-radial seals are the best ones in terms of vibration stability and durability.

(2) Thoughtless replacement of conventional and the more so axial-radial seals by honeycomb ones is inadmissible, especially in T-250/300-23.5 turbines featuring poor vibration stability of the shaft system. With such replacement, the conditions of calculation-and-design, installation, and operation nature outlined in the article must be complied with; in particular, self-aligning peripheral seals must be developed and used.

(3) The use of variable-pitch multicomb seals, which differ advantageously from the other ones in economic efficiency, vibration stability, durability, and cost, is a good alternative to overshroud honeycomb seals.

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